Heat Exchanger Thermal Design of Oil System for Turbo Centrifugal Compressor Using Nanofluid

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Abstract— *A thermal management is vital issues of all energy equipment such as compressor, gas turbine, and boilers etc. The compressor is generally used in power, oil & gas, air separation, and chemical plant. It is consist of air or gas compression part, gear, bearing, cooling, sealing, lube oil, and control system.*

In this study focused on heat exchanger for oil supply systems. Lube oil is very important to supply oil and protect bearing. Lube oil's temperature control is vital issue to prevent system broken. Shell and tube heat exchanger is used as a cooler.

In this study, HTRI Xist used to thermal design of oil cooler, with water and nanofluid. The thermal conductivity is ~9.3% higher than water. The tube side overall heat transfer coefficient of nanofluid is increased by ~9% compared to that of water.

Keywords— HTRI, Nanaofluid, Shell and tube heat exchanger, oil system cooling, Turbo centrifugal geared compressor

I. INTRODUCTION

A thermal management is essential to improve system's stability and performance of energy equipment. A compressor, gas turbine, and boilers are widely used as an energy equipment. The compressor consist of air compression, gear, bearing, sealing, cooling, lube oil, and control system. Compressor package shall include, integrally geared centrifugal air compressor, coupling and coupling guard, baseplate, intercoolers and aftercooler, moisture separator and drain system, lubrication oil system, controls and instrumentation, driver, interstage air piping, inlet and discharge expansion joints and assessories as API 672 packaged, integrally geared centrifugal air compressors for petroleum, chemical, and gas industry [1]. Among them lube oil system for a machine must provide with proper amount and control the temperature [2]. High performance cooling is vital needs in lube oil system in order to thermal management. However, the thermal conductivity of conventional coolants such as water, ethyleneglycol, propyleneglycol, and/or their mixture is fixed and difficult to achieve high performance cooling [3]. It is possible to increase the heat remove by increasing the contact area between the coolant and the heating surface. However, the current design has already adopted the extended surface technology to its limits. Generally, the advanced heat transfer given by [4]

$$h = \frac{Nu}{D} k_f$$

Where, h is heat transfer coefficient, Nu is the theory of conventional heat transfer, which is Nusselt number (h_x/k), k_f is thermal conductivity of coolant, and D is system size which means heating surface area.

Further, heat flux (q_x) is can be described as [5]

$$q_x = -k\frac{dT}{dx}$$

Where, k is thermal conductivity of coolant, dT is temperature difference, and dx is size difference which means heating surface area.

From any heat related formula, it is considerable way to improved heat transfer coefficient (cooling capability) is increasing the thermal conductivity of coolant. If the thermal conductivity of coolant can increase, it is possible to obtain much higher heat transfer coefficient or heat flux without any size change. At present, many researchers have come up with a way to change the thermal properties of coolant [6-9].

Current nanotechnology can provide wonderful opportunity of materials with sizes nanomaters level. Recognizing an opportunity to apply this nanotechnology to be established thermal management, Masuda group reported that a suspension of ultra-fine particles in a basefluid (water) enhances the thermal conductivity of the basefluid [10]. Also Choi proposed that nanometer sized particles suspended in the basefluid, which is so called "nanofluid", to produce a new class of engineered fluid with high thermal conductivity [11].

On the other hands, many researchers have been reported heat transfer characterization of shell and tube heat exchangers [12-14]. However, the effectiveness of nanofluid in compression oil system cooling as an oilcooler and its design approach is lack.

In this study, 3 stages turbo centrifugal compressor designed using S. Hall's equation [15]. It calculated their mechanical loss which means oil system requirement and oil cooler designed using HTRI (heat transfer research institute Inc.). HTRI has developed powerful thermal design and rating tool of commercial heat exchanger such as shell and tube, hair pin, plate and frame, spiral plate and aircooled etc. The nanofluid fabricated by pulsed laser ablation (PLAL) method as a single-step method [7]. The thermal conductivity measured by transient hot-wire method [16]. The oilcooler designed using Kern's equation [17] and HTRI.

II. THERMAL DESIGN

2.1 Compressor design and oil system requirement

Table 1 summarized compressor design requirement and table 2 summarized supply cooling system information.

Table.1: Compressor design requirement

Item		Content	Unit
Fluid		Air	-
Suction	Flow rate	1.50	kg/s
	Temperature	20.00	C
	Pressure	1.013	barA
Discharge	Temperature	42.00	C
	Pressure	6.70	barA
Humidity		60.00	%

Table.2: Supply cooling system

Item		Content	Unit
Fluid		Water Nanofluid	-
Inlet	Temperature	32.00	C
	Pressure	5.00	barA
Outlet	Temperature	42.00	C

Based on given system design requirement and information, the compressor designed. Table 3 shows design result by aero power, mechanical loss and total power. This system required 10.56kW level oil supply system. Normally it considered design and system margin 10%, so the oil cooler's heat exchanged (duty) should be 11.6kW.

Table.3: Compressor design result

Polytropic aero-power	80.62	kW
Mechanical loss	10.56	kW
Total power	372.88	kW

2.2 Coolant thermal conductivity

Fe_2O_3 nanofluid fabricated by pulsed laser ablation in liquids (PLAL) method as a single-step method [7]. A Q-switched Nd:YAG laser was used to produce Fe_2O_3 nanofluid by varying an irradiation time 18hours. Fig. 1 shows the thermal conductivity of basefluid and nanofluid. The thermal conductivity of the nanofluid increased by ~9.3% compared to that of basefluid.

HTRI has simple material input panel. In HTRI using HTRI own material data as a material package. However, it is possible to linked property package such as VMG Thermo, HYSYS, REFPROP, and CAPE-OPEN.

In this study using measured coolant thermal conductivity as a material input. Basefluid material package based on HTRI database.

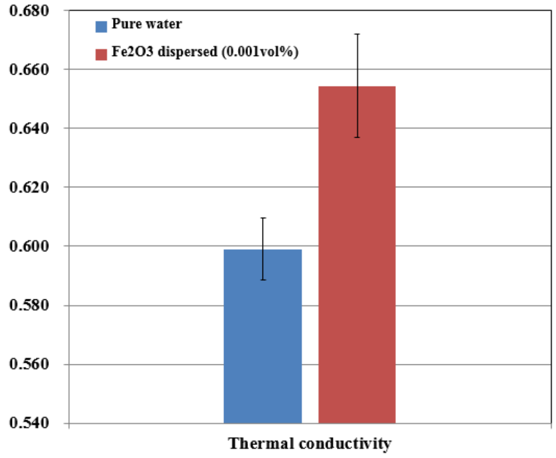


Fig. 1: Thermal conductivity of basefluid and nanofluid

2.3 Oil cooler design requirement

Petroleum, chemical, and gas industry has required apply API standard [1]. Table 4 summarized cooling system requirement by API 672 [1].

Table.4: Heat exchanger design requirement

Item		Content	Unit
Fouling resistance		0.00018	m^2K/W
Max. pressure drop		1.00	bar
TEMA class		C	-
Tube side	Material	Cu 90/10 or Cl220 Cu	-
	Out-diameter	Not less than 15	mm
	Wall-thickness	1.25	mm

Fig. 2 shows oil cooler design input condition as HTRI input. Shell type is BEM and shell inner diameter is 280mm with single pass. Tube type is plain with single segmental perpendicular oriented baffle, tube material is Cu. Tube out-diameter is 15.875mm, wall-thickness is 1.245mm with 30 degree layout angle (19.844mm of pitch) and 6 tube passes. Tube count is 52 and tube length is 1.4m.

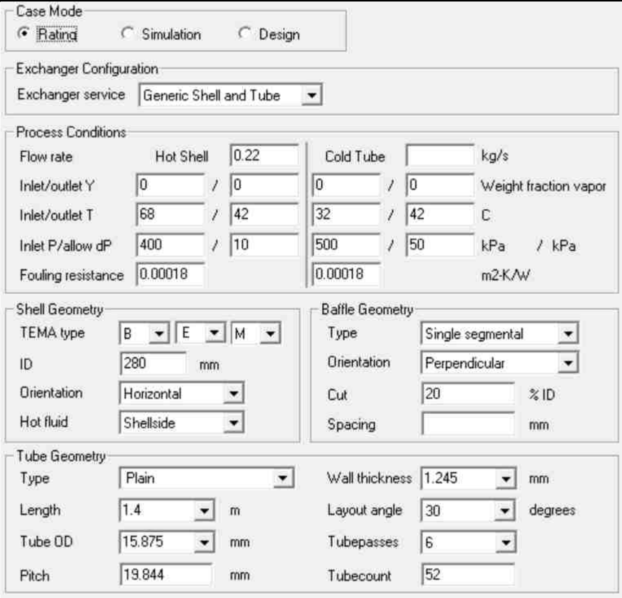


Fig.2: HTRI input condition

HTRI has 3 design modes. There are rating, simulation, and design. Among them most thermal engineer using rating mode cause checking design margin based their input size and design requirement and quick design checking [14]. In this study, using rating mode.

Fig. 2 shows oil cooler's 2D sketch layout with in/out nozzle of shell and tube side. Fig.3 shows oil cooler's 3D layout with outer and inner side from HTRI result.

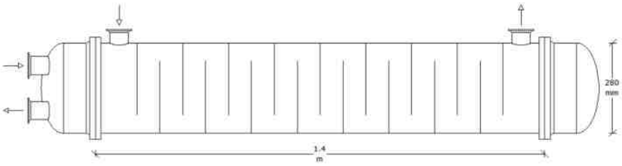
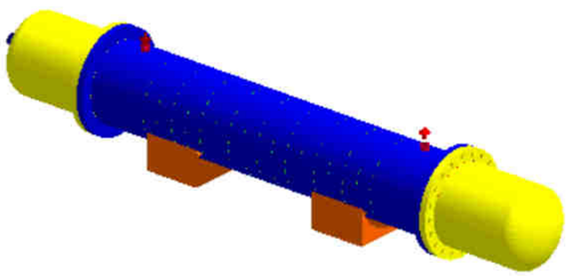


Fig.2: Oil cooler 2D sketch layout



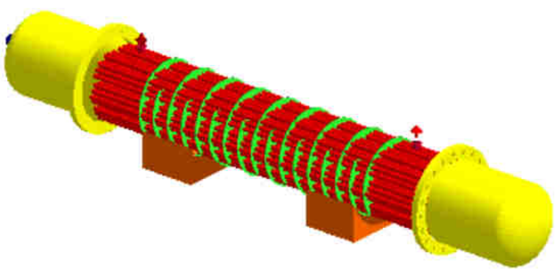


Fig.3: Oil cooler 3D layout outside and inside

III. TERMAL DESIGN RESULT

Table 5 shows the summarized of thermal design result of oil cooler by Kern's equation. As shown in Table 5, overall heat transfer enhanced by ~9% and tube side heat transfer enhanced by ~7%. However, there are no change in shell side heat transfer. We assumed, if *Ru* is not chagned there are no change the heat transfer coefficient. In shell side there no changes the Re and Pr both result of kern's equation and HTRI result. However, it is dramatically changed tube side heat transfer capability.

Table.5:Thermal design result by Kern's equation

Item		Basefluid	Nanofluid	Unit
Heat load		11.9	11.9	kW
Area		2.9	2.4	m^2
Overall *U*		219	241	W/m^2K
Tube	Re	3452	3464	-
	Pr	4.43	3.99	-
	U	1061	1139	W/m^2K
Shell	Re	621	621	-
	Pr	15.83	15.83	-
	U	314	314	W/m^2K

Table 6 shows the summarized of thermal design result of oil cooler by HTRI. As shown in Table 6, overall heat transfer enhanced by ~9% and tube side heat transfer enhanced by ~8%. However, there are no change in shell side heat transfer as same with Kern's equation.

Table.6: Thermal design result by HTRI

Item		Basefluid	Nanofluid	Unit
Heat load		11.6	11.6	kW
Area		3.5	3.5	m^2
Overall *U*		222	245	W/m^2K
Tube	Re	4476	4563	-
	Pr	4.54	4.00	-
	U	1256	1359	W/m^2K
Shell	Re	1164	1164	-
	Pr	19.54	19.54	-
	U	356	357	W/m^2K

IV. DISCUSSION

To understand the effectiveness of nanofluid in heat exchangers, compared heat transfer coefficient of overall, tube, and shell side, calculated *Re*, and *Pr*.

Heat transfer coefficient equation is below [17].

$$\frac{1}{U} - \left(\frac{1}{h_t}\right) + \left(\frac{1}{h_s}\right) + \left[\left(\left(tubeOD \times \ln\left(\frac{tubeOD}{tubeID}\right)\right)/(2 \times k_{tube})\right) + \left(\left(\frac{tubeOD}{tubeID}\right) \times (2 \times flowrate)\right)\right]$$

Where, U is overall heat transfer coefficient, h_t and h_s is heat transfer coefficient of tube and shell side, respectively, k_{tube} is thermal conductivity of tube material.

Reynolds number Prandtl number equation is below [17].

$$Re = \frac{(\rho * u * di)}{\mu} \qquad Pr = \frac{(C * \mu)}{k}$$

Where, ρ is density, u is linear velocity, di is tube inner diameter, μ is dyanmics viscosity, C is specific heat capacity, k is thermal conductivity.

Fig. 4 shows tube side heat transfer coefficient comparison. Usign nanofluid the heat transfer coeffient is increased by 7~8% compared to that of basefluid. However, as shown in Fig. 5, shell side heat transfer coefficient was not changed cause shell side property is unchanged.

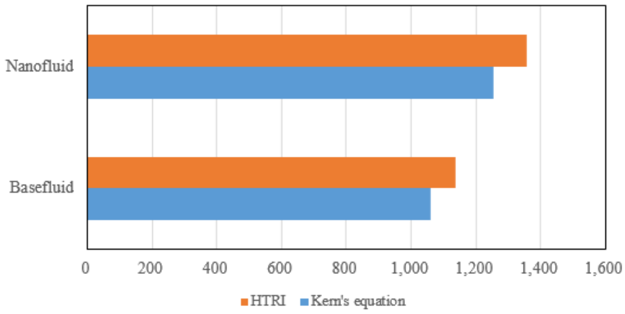


Fig. 4: Tube side heat transfer coefficient comparison

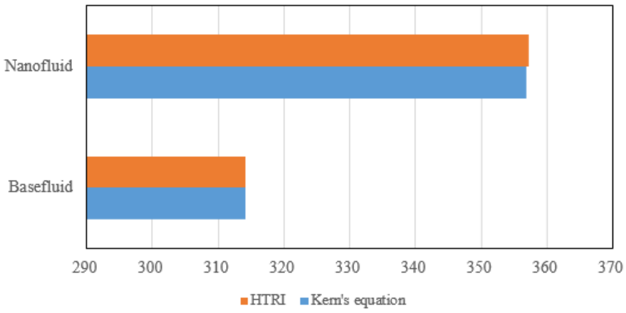


Fig. 5: Shell side heat transfer coefficient comparison

Fig. 6 shows overall heat transfer coefficient comparison. *Pr* is decreased by 11~13% compared to that of basefluid *Pr* is dimensionless number defined as the ratio of momentum diffusivity to thermal diffusivity [18]. Small values of *Pr* means the thermal diffusivity dominates, the other hands large values of *Pr* means momentum diffusivity dominates it behavior. Decreased by 11~13% of nanofluid, it means that the heat diffuses quickly compared to the velocity. This means that for nanofluid thickness of thermal layer is larger than basefluid. It is clearly match to increased thermal conductivity of nanofluid [14].

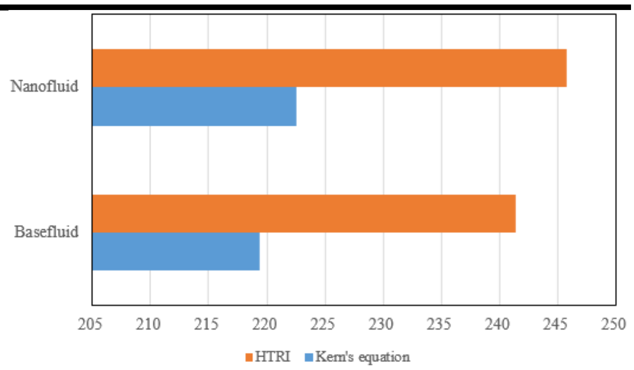


Fig. 6: Overall heat transfer coefficient comparison

Comparison of Kern's equation and HTRI regarding to overall heat transfer coefficient, tube side *Re* and *Pr* different prediction. This is variation in kern's equation and HTRI rating logic may be due to different calculation of *Pr* and *Nu* based on material property. HTRI has been calculation their material database not only thermal conductivity but also viscosity etc.

V. CONCLUSION

We have thermal design characteristics of heat exchanger for compression oil system cooling. The overall heat transfer coefficient of heat exchanger using nanofluid was increased by ~9% compared to that of basefluid. Tube side *Pr* number of heat exchanger using nanofluid was decreased by 11~13% compared to that of basefluid.

We believe that the present design approach is a useful for compression oil system cooling design. More research will be carried out to find thermal effectiveness of nanofluid in order to understanding the heat transfer mechanism.

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